TECHNICAL NOTES

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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RISE IN TEMPERATURE OF THE CHARGE IN ITS PASSAGE THROUGH THE INLET VALVE AND PORT OF AN AIR-COOLED AIRCRAFT ENGINE CYLINDER

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Washington
January 1942

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SUMMARY

The heat transfer between the air stream and a model of the inlet valve and seat of an air-cooled aircraft type cylinder (Wright J-6) has been experimentally determined as a function of air flow and valve lift. Inlet valve and seat temperatures and air consumptions have been determined experimentally in a single-cylinder engine under operating conditions. The inlet port of the flow model was cut from a cylinder of the same design as the one operated in this series of tests. Calculations of the heat transferred to the fresh charge from the inlet valve and seat under actual operating conditions of the engine have been made by use of data obtained from experiments with the flow model. The effect of inlet valve and seat cooling on volumetric efficiency has been determined experimentally.

The over-all temperature rise of the charge up to the point of inlet-valve closing has been computed from experiments on the engine under normal operating conditions. It was found that about 35 percent of this temperature rise was due to heat transferred from the inlet valve and seat.

A general formula based on thermodynamic analysis is derived for the over-all temperature rise of the charge prior to the inlet valve closing. Application of this analysis to light-spring indicator diagrams taken on this engine shows that as much as 30 percent of the total temperature rise was the result of inlet-valve flow resistance.

INTRODUCTION

In a previous report (reference 1) the relative importance of inertia and friction in controlling the quantity of air flowing into an engine cylinder was investigated with a view to explaining the volumetric deficiency, that is, the difference between the volumetric efficiency and unity. The results showed the pressure drop in the valve to be responsible for an extremely small part and the rise in temperature of the charge before inlet closing to be responsible for a relatively large part of the volumetric deficiency.

It is the purpose of the present investigation to make a quantitative study of the effects of heat transfer and pressure drop in the inlet valve of an air-cooled engine and to correlate these effects with the observed volumetric deficiency.

APPARATUS AND PROCEDURE

Flow Model and Tests

In order to obtain heat-transfer data pertaining to the flow of charge through the intake port of the actual engine, a section comprising the intake port, seat, and portion of the cylinder was cut from a Wright J-6 cylinder. (See fig. 1.) A jacket of sheet brass around this cutout section made it possible to control the temperature of the seat, port, and surrounding cylinder portions by the introduction of water heated by steam.

A model hollow-stem valve similar in design to that employed in the engine was used in this model. (See fig. 2.) This valve was made of high conductivity material (beryllium copper) to insure that its surface was at nearly uniform temperature. The temperature of the valve was controlled by regulating the temperature of water circulating in the hollow stem. The valve lift was controlled by a micrometer screw. Temperatures of the seat and the valve were measured by iron-constantan thermocouples.

By means of a Nash pump, air was drawn successively through an air-measuring orifice and through the flow model. In order to reduce errors in measurement the flow model was insulated from the inlet and the discharge pipes by short sections of rubber hose. Thermometers at the inlet and the outlet ends of the model allowed measurement of the rise in the temperature of the air due to heat transferred from the model. A manometer indicated the pressure drop across the valve.

Temperature-rise ratios for various flow quantities and valve lifts were obtained on the flow model in the following manner: For a given setting of valve lift and air-flow rate, the temperatures of the valve and seat were equalized by suitable regulation of the heating steam and water. When the valve-and-seat temperature had attained equilibrium, the inlet and the outlet temperatures of the air were noted. This procedure was carried out at various valve lifts and air quantities.

The temperature-rise ratio Φ_a is defined as the ratio of the difference between the outlet-air and the inlet-air temperatures to the difference between the valve-and-seat temperature and the inlet-air temperature. Values of Φ_a were plotted against air flow at various valve lifts. (See fig. 3.) It is notable that these curves of temperature-rise ratio against air quantity for valve lifts greater than about 0.1250 inch are practically coincident. The spread of points in figure 3 at the higher air quantities is quite marked. Fortunately, it was unnecessary to use this region of the curves in the calculations for the actual engine.

A series of runs on the flow model was made in which the drop in pressure across the valve at various constant valve lifts was determined as a function of flow quantity. The valve, the valve seat, the port, and the flowing air were at room temperature throughout these runs. A check on the validity of the assumption that the relation between air flow and pressure drop was independent of port and valve temperature was carried out for the highest flow-model temperatures, and the assumption was found to be justified. Curves of the square of the air flow against pressure drop at various constant valve lifts are given in figure 4.

Modified Engine and Tests

The engine used in this investigation was a modified Wright J-6 air-cooled cylinder of 5.00 inches bore and

5.50 inches stroke mounted on a universal test crankcase. The compression ratio was 5.1. A 6-volt ignition system with one spark plug was used.

Valve modifications and cooling.- The modifications (figs. 2 and 5) consisted of boring out the intake-valve guide boss to take a standard exhaust-valve guide and the installation of a valve-seat insert with an annular passage for the circulation of cooling water. Drilled passages led from the outside of the cylinder to the cooling passage around the seat.

A modified hollow-stem exhaust valve was used as the intake valve. The modification (fig. 2) consisted of arranging the valve stem with an inner tube and connections so that water could be circulated continuously through the interior of the stem and the head. During engine operation water was introduced and removed from the valve by rubber tubes (see figs. 2 and 5) attached to the fixture interposed between the end of the valve stem and the valve-operating plunger. A tightly fitting rubber tube extending between the fixture and the valve stem prevented cooling-water leakage at this junction. By means of these arrangements either valve or seat or both could be cooled in varying degrees.

Inlet system.- Air was supplied to the engine by a Nash pump, passing in turn through a standard air meter, a surge tank, a throttle valve, a fuel-mixing orifice, and a vaporizing tank.

Air was measured by a 1-inch-diameter sharp-edge orifice, designed in accordance with the A.S.M.E. specifications (reference 2). The downstream side of the orifice was connected to the top of a 50-gallon surge tank by a short section of 3-inch pipe. Air from the surge tank passed through a gate valve and a fuel-mixing orifice to a jacketed tank of 14 gallons capacity. This tank served as a combined vaporizing and auxiliary surge tank. The incoming air from the gate valve was mixed at the fuel-mixing orifice with the spray from a Bosch fuel-injection pump. The vaporizing tank was connected to the inlet port of the engine by means of a 3-foot length of 2-inch pipe. A thermometer was located in this pipe about 1.5 feet from the inlet port. The inlet temperature at this point was maintained constant by adjusting the supply of steam and water to the jacket on the vaporizing tank.

Pressure in the vaporizing tank was maintained constant by adjustment of bleed valves on the upstream side of the measuring orifice.

Exhaust system.- A surge tank of about 10 gallons capacity was connected by a 1.5-foot length of 2-inch pipe to the exhaust port of the engine and thence connected to the exhaust mains by means of a gate valve. This valve enables the pressure in the tank to be maintained at any constant level. A Cambridge exhaust-gas analyzer and a water manometer were attached to this tank.

Engine instruments.- Engine speed was controlled by a combination of a conventional tachometer and a stroboscopic light running directly from the 60-cycle alternating-current supply that illuminated painted strips on the flywheel. This method of control is described in reference 3. A Sprague electric cradle dynamometer was used. Pressure against crank-angle diagrams were obtained from the standard M.I.T. balanced-pressure indicator using M.I.T. "flapper valve" and diaphragm pressure units (references 1 and 4).

Temperature Measurements

Four holes were drilled through the valve head on a circumference about midway between those of the seat and the stem, and into these holes were brazed two rods of constantan and two of iron about 0.065 inch in diameter and 0.75 inch long. A detail of one of these connections is shown in figure 2. The intervening steel of the valve combined with any two dissimilar rods formed a conventional iron-constantan thermocouple. In order to make contact with these rods, flexible insulated leads that would stand up under the valve motion were required. After several attempts had been made to use sliding contacts, positive-contact flexible leads were developed. These leads were made by closely winding No. 28 cotton-covered iron or constantan wire on a 0.C55-inch diameter mandrel and then sliding tightly fitting insulating spaghetti tubing over this coil. After the spaghetti tubing had been fitted, the mandrel was pulled out. Since this sheathed coiled conductor had a tendency, if bent, to hinge at some particular point, a snugly fitting coiled spring of 0.014-inch-diameter piano wire was made to slip over the outside of the spaghetti tubing. This spring had the property of distributing the bending, and the close coiling of the thermocouple wire prevented localized high stresses therein, while at the same time the whole assembly was reasonably flexible.

Owing to the fact that the internal diameter of this flexible lead was less than that of a rod welded to the valve head, it was possible to force it over the entire length of the rod and then carry its free end to a similar rod and weld it thereon. This method of securing the leads prevented any weakening of the coil from brazing temperatures. The ends of these two leads were brought through conventional airtight packing glands in the intake port wall, leaving the piano-wire coils under compression along the length of the spaghetti tubing between the valve head and the packing glands.

Iron-constantan leads from a thermocouple embedded in the valve seat were also brought out through packing glands in the inlet port. A standard iron-constantan spark-plug-gasket thermocouple was used to indicate cylinder temperature. Cylinder, valve, and valve-seat temperatures were measured on a direct-reading potentiometer. All other temperatures were measured with mercury-in-glass thermometers. The cylinder temperature as indicated by the spark-plug-gasket thermocouple was controlled by regulating the speed of a motor-driven centrifugal blower that forced air through a duct into the cylinder cowling.

SYMBOLS

Φ temperature-rise ratio (ratio of difference between outlet working fluid and inlet working fluid temperature to difference between valve and seat temperature and inlet working fluid temperature)

Q quantity of heat, Btu

Q_T total heat, Btu

Q_{ec} heat transferred to charge during interval e to c

H_i enthalpy of fresh charge in inlet manifold

H_c enthalpy of fresh charge at point c at beginning of compression stroke after closing of inlet valve

H_{cf} enthalpy of fresh charge at point c

H_{cr} enthalpy of residual gas at point c

θ crank angle, degrees

T absolute temperature (460+ °T)

t temperature, °F

t_p cylinder-head temperature, °F

t_v intake-valve temperature, °F

t_s intake-valve seat temperature, °F

t_{vs} average temperature of intake valve and seat, °F

$\Delta_1 t$ $t_{vs} - t_i$

$\Delta_2 t$ $\Phi' \Delta_1 t$

M weight of working fluid delivered to cylinder, pounds per stroke

M_m weight of fresh mixture delivered to cylinder, pounds per stroke $\left[M_a\left(1 + \frac{F}{A}\right)\right.$, where $\frac{F}{A}$ is fuel-air ratio of mixture$\left.\right]$

M_f weight of air in fresh charge

M_r weight of air originally in residual gas before burning

w weight rate of flow of working fluid, pounds per second

q quantity of air discharged through valve, pounds

(When w, q, and Φ are primed, they refer to weighted values)

c_p specific heat at constant pressure

e volumetric efficiency

n engine speed, revolutions per second

V_{er} specific volume of residual gas at point e on indicator card

V_{cr} specific volume of residual gas at point c on indicator card

V_{cf} specific volume of fresh charge at point c on indicator card

V_c specific volume of charge at point c

v_1 clearance volume

v_{er} volume occupied by residual gas at point e; that is, v_1

v_c cylinder volume at bottom center

v_{cr} volume occupied by residual gas at bottom center

v_d piston displacement

E_c internal energy of charge at point c

E_{cf} internal energy of fresh charge at point c

E_r internal energy of residual gas at end of exhaust stroke

E_{cr} internal energy of residual gas at point c

J mechanical equivalent of heat

P_e pressure in cylinder at end of exhaust stroke

P_{er} pressure of residual gas in cylinder at end of exhaust stroke

P_c pressure in cylinder as noted on indicator card at point P_c

p_{cr} pressure of residual gas at point c

p pressure in cylinder

f proportion of residual gas

k ratio of specific heats

ρ density of working fluid, pounds per cubic foot

ρ_i density of fresh charge in inlet manifold

ρ_c density of fresh charge at point c in cylinder

Subscripts:

a fresh air

m fresh mixture

f fresh charge

r residual gas

c fresh charge at point c in cylinder before mixing (see figs. 9(a), 9(b))

e point e on indicator cards (figs. 9(a), 9(b))

i inlet

These quantities for which the thermodynamic properties of the working fluid are computed are the same as used in Hottel charts (reference 5), that is, 1 pound of air.

RESULTS AND DISCUSSION

Heat Taken Up by the Charge from Seat and Valve

If the temperature-rise ratio between the air stream and the inlet valve and seat of an engine cylinder be known as a function of valve lift and rate of flow of charge and, furthermore, if the valve and valve-seat temperatures together with the inlet temperature be known, it is possible to estimate the quantity of heat per stroke taken up by the charge from the seat and the valve.

The steps leading to the evaluation of this quantity are as follows.

The intake-valve lift was determined as a function of crank angle by means of a dial indicator when the engine was cold. The valve lift against crank-angle curve corrected for the difference between the hot and cold clearances is shown in figure 6.

Under running conditions a light-spring indicator diagram yields the relationship between cylinder pressure and crank angle. A second light-spring indicator diagram of the intake-port pressure yields a relationship between crank angle and intake pressure. From these two diagrams a curve of valve-pressure drop against crank angle can be constructed. (See fig. 6.) The relationship between valve pressure drop and rate of flow through the valve at various lifts is known from the flow-model experiments on an identical valve and port. (See fig. 4.) Thus, the approximate instantaneous rates of flow through the valve under running conditions can be obtained by assuming that the rate of flow in operating is the same as the steady rate of flow for a constant pressure drop. Reference 1 indicates that the magnitude of the error introduced by this assumption is not large. It is now possible to perform a step-by-step integration of the quantity of heat taken up by the charge during all phases of the open-valve period, as detailed in the following paragraphs.

About some particular valve lift, say 0.0625 inch, a small crank-angle interval extending on both sides of the lift is chosen. For this crank-angle interval an average value of the corresponding valve-pressure drop can be obtained from the valve-pressure against crank-angle curve. This average valve-pressure drop in turn yields an average flow quantity from the flow-quantity against valve-pressure-drop curve corresponding to the 0.0625-inch lift curve. The average flow quantity then yields an average temperature-rise ratio from the corresponding 0.0625-inch lift curves of flow quantity against temperature-rise ratio. In appendix I it is shown that the temperature-rise ratio of the fresh charge is 99.7 percent that of air. Values of the temperature-rise ratio taken from figure 3 for use in engine calculations have been corrected by this amount.

The average temperature-rise ratio for this crank-angle interval and the temperatures of the valve,* the seat, and the charge at the inlet being known, an average temperature rise of the charge can be assigned to this interval. Since in the engine the valve and seat temperatures were not the same (see tables III and IV), their arithmetic mean was used in computations. Results obtained by the use of weighted average temperatures of the valve and seat gave no better correlation than the results obtained from the arithmetic mean temperature. From the known engine speed, the time corresponding to the crank interval is calculable. The product of the time interval, the average quantity rate, and the average temperature rise, when multiplied by the specific heat of the charge mixture, yields the average heat taken up by this fraction of the charge.

Carrying through the foregoing process for a number of steps including the entire period of valve opening and summing the result yields an approximation of the heat transfer to the charge per inlet stroke. Table I gives the results of such an integration.

The work involved in integrating the heat Q picked up from the valve and the seat is reduced to a comparatively brief calculation through the following simplification.

For any particular run the intake-air meter provides a measure of the charge taken in per stroke. The complete open-valve time interval is provided by the valve-lift against crank-angle curve and, if this interval is divided into the total charge per stroke, there results an average rate of flow of charge into the cylinder. For this average rate a corresponding temperature-rise ratio can be taken from the curve (fig. 3) representing the higher lifts. This temperature-rise ratio ϕ' when multiplied into the difference between the average seat and valve temperature and that of the inlet yields a value of the average temperature rise of the charge while flowing by the seat and the valve. As before, multiplying the charge per stroke into the product of this average temperature rise and the specific heat results in a measure of the heat Q picked up by the charge in transit through the port into the cylinder.

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*The valve used in the engine had a very low thermal conductivity, which made it impossible to reduce the temperature of the valve head at the point where the thermocouple was attached to the temperature of the seat even though water was circulated through the valve so rapidly that it came out cold.

The results of integrating eight different runs compared with those obtained on the same data by this short-cut method show a maximum divergence of only about 3.5 percent. (See table II.) All the values of Q submitted in this report (table III) are based on this short-cut method of calculation.

Heat Taken Up before Valve Opens and after Charge Enters Cylinder

The foregoing process does not account for the heat picked up by the charge before the valve opens or after the charge enters the cylinder.

An estimate of the temperature of the charge in the cylinder before compression, and hence the total heat transfer to the charge, can be obtained from the measured air consumption in the following manner.

The volumetric efficiency of the engine is defined by

$$e = \frac{w_m}{\frac{n}{2} v_d \rho_i} \qquad (1)$$

The weight rate of flow of charge w_m is determined from the intake-air meter and the fuel-air ratio. It is measured in pounds per second. For the air-fuel ratio used in this series of runs (A/F = 12.2)

$$\rho_i = 1.41 \frac{P_i}{T_i} \qquad (2)$$

The inlet pressure P_i was measured by a manometer attached to the surge tank. It is given in inches of mercury. The inlet temperature T_i was measured in the inlet pipe about 1.5 feet from the inlet port. It is given as absolute temperature.

The induction process may be considered to be divided into two steps:

(a) Fresh charge flowing into the cylinder without mixing or exchanging heat with the residual gas. Reference 1 indicates that the magnitude of the error introduced by this assumption is small.

(b) Mixing of residual and fresh charge.

The subscript c refers to the state of the fresh charge after the completion of the induction and before the mixing process. At this time the residual gas is assumed to occupy the clearance volume. This assumption is justified since the pressure P_c differs little from the pressure at top center on the exhaust stroke; consequently, there is little net expansion or compression of the residual gas. It follows that the volume occupied by fresh charge is v_d, the displacement volume. The determination of P_c is discussed in greater detail in a later section.

The weight of charge delivered to the engine per stroke can be calculated from W_a, the measured rate of air flow to the engine in pounds per second, the air-fuel ratio A/F, and n, the revolutions per second of the engine. According to step (a) this value will also be equal to $(\rho_c v_d)$. It follows then by definition that the volumetric efficiency is

$$e = \frac{\rho_c v_d}{\rho_i v_d} = \frac{\rho_c}{\rho_i} \tag{3}$$

The ratio ρ_c/ρ_i is, by the gas laws, equal to $\dfrac{P_c}{P_i}\dfrac{T_i}{T_c}$ and therefore

$$e = \frac{P_c}{P_i}\frac{T_i}{T_c} \tag{4}$$

Figure 7 shows the variation of the volumetric efficiency with average valve and seat temperature at the two speeds.

By means of equations (1) and (4) it is possible to solve for T_c, the temperature of the total charge just prior to compression, since all the other variables are known or can be calculated.

The total rise in temperature of the charge $T_c - T_i$, multiplied into the product of weight of the charge and specific heat of the charge, measures the over-all heat Q_T picked up by the charge in its passage from the inlet until it arrives in the cylinder and the inlet valve closes. The relative contribution of the valve and seat to the over-all heat picked up by the charge in its passage from the intake manifold until it is compressed can be estimated from the foregoing quantity. Experimental values of Q_T together with the factors leading to its evaluation are shown in table IV.

Figure 8 shows the ratio of heat transferred from the inlet valve and seat to the over-all heat picked up by the charge as a function of the average valve-and-seat temperature.

An appreciable part of the total heat picked up by the charge during the suction stroke can be attributed to work done in overcoming the pressure drop in the valve. This quantity may be estimated as follows.

The following equation can be derived from the first law of thermodynamics,

$$(M_f + M_r)E_c - M_f H_i - M_r E_r = -\frac{1}{J}\int_e^c P dv + Q_{ec} \qquad (5)$$

This equation may be used directly to give the unknown quantity E_c and hence the temperature T_c. In this case, however, it is desirable to know what contributions to the temperature T_c are made by heat transfer, by the work done by the piston during the suction process, and by the heat interchange between the fresh charge and the residual gas. The last of these quantities may be estimated by imagining that the residual gas is left separated from the fresh charge by an insulating membrane until after the point c so that the pressure of the residual gas is always the same as the pressure of the remainder of the cylinder contents and the residual gas is under adiabatic conditions. The usual laws of perfect gases are assumed to hold because of the relatively low temperatures involved in the induction process. With these assumptions equation (5) may be rewritten as follows. (See appendix II for derivation.)

$$T_{cf} - T_i = \frac{Q_{ec}}{c_p(1 + f)M} + \frac{1}{c_pJ(1 + f)M}\left[(P_cv_c - P_cv_i) - \int_e^c P dv\right]$$

$$+ \frac{v_i}{c_pJ(1 + f)M}\left\{P_c\left[1 - \left(\frac{P_{er}}{P_c}\right)^{\frac{1}{k}}\right] + \frac{P_{er}}{k - 1}\left[1 - \left(\frac{P_c}{P_{er}}\right)^{\frac{k-1}{k}}\right]\right\} \quad (5a)$$

The right-hand side of equation (5a) now consists of three terms, the first of which gives the change in temperature of the fresh charge due to heat transfer. If $P_c = P_e$, the third term is zero and the second term represents the change in temperature of the fresh charge due to work done on it by the piston and by the gas remaining in the inlet manifold. The bracketed part of the second term of equation (5a) is also equal to the shaded work area shown on the light-spring indicator cards in figures 9(a) and 9(b). It is therefore possible to make a rapid estimate of the rise in temperature due to work done while drawing the charge through the resistance of the inlet valve, provided that $P_c = P_e$. Even if P_c is not exactly equal to P_e, the third term in equation (5a) is small; in fact, it was found to be negligible for the cases herein investigated.

The effect of mixing of residual gas and fresh charge is analyzed in reference 1. The analysis in this reference showed that, for a compression ratio of 6.5, the mixing of the fresh charge with the residuals decreased the volumetric efficiency about 1.5 percent. It is therefore roughly equivalent to heating the fresh charge by this amount, or about 10° F.

Determination of Point c

The point c should be taken after closing of the inlet valve in order for equations (5) and (5a) to be valid. On the other hand, if the point c is taken some distance up the compression curve, the second term of equation (5a) will include some work of compression as well as the work done while drawing in the charge. This difficulty was solved by measuring the pressure of the charge at several points along the compression line and assuming reexpansion of the charge to the cylinder volume at bottom center. The resulting pressures were averaged to give a hypothetical pressure P_c that, with the cylinder volume at bottom center and a known weight of charge, determined the conditions at point c.

Value of Temperature Rise Attributable to Flow Resistance.

Measurement of the shaded area of the diagrams of figures 9(a) and 9(b) indicated that the temperature rise attributable to flow resistance in the inlet valve was 26° F at 1500 rpm and 12° F at 1000 rpm. The rapid increase in temperature rise with engine speed is to be expected; the result indicates that the rise in temperature attributable to flow resistance is a much more important factor at high piston speed than at low piston speeds and may be to a considerable extent responsible for the falling off of indicated mean effective pressure at high piston speeds.

Heat Transfer to Charge before Inlet Valve Opening

If the total heat gained by transfer and work of compression is subtracted from the total heat picked up by the charge, the result is a measure of the unaccounted-for heat picked up while the charge is behind the closed valve and after it has entered the cylinder. While the charge is at rest in the inlet pipe, a part of the charge is in a region where the wall temperature is high. If it is assumed that a section of the air in the inlet pipe (3 in. out of a total of 37 in. occupied by the charge) attains the average temperature of the seat and the valve before the inlet valve opens while the remainder is at inlet temperature, heat so computed is about equal to the difference between the total heat and that accounted for elsewhere. This illustration gives an idea of the possible contribution to the total heat transfer from this source.

An upper limit for the sum of the temperature rise due to flow resistance and that due to transfer after the charge has entered the cylinder can be obtained by extrapolating the value of volumetric efficiency in figure 7 to an average valve-and-seat temperature equal to the inlet temperature and inserting this value in equation (4). These limits have been established as 51° F and 57° F at 1500 rpm and 1000 rpm, respectively.

Importance of Low Valve-and-Seat Temperatures

From experience with many engines, it is found that a 10° F change in inlet temperature results in a change in air capacity that indicates that the charge temperature is changed only 5° F. (This fact is the reason for the success of the square-root rule in correcting indicated horsepower, as explained in reference 6.) The results of the present investigation show that a 5° F decrease of charge temperature can be obtained by lowering the average inlet valve-and-seat temperature 45° F. Rothrock and Biermann show (reference 7) that a decrease of 10° F in inlet temperature permits an increase of inlet pressure to obtain an increase in indicated mean effective pressure of about 3.2 percent, keeping the tendency to detonate constant. A decrease in average valve-and-seat temperature of 45° F, which results in the same reduction in charge temperature as a 10° F reduction in inlet temperature, should give the same permissible increase in mean effective pressure. Thus a 10° F reduction in inlet valve-and-seat temperature should permit an increase of approximately 0.7 percent in indicated horsepower without increasing the tendency to detonate.

CONCLUSIONS

1. In reference 1 it was noted that the pressure in the cylinder at the end of the suction stroke was practically the same as the pressure in the manifold; hence the volumetric deficiency, that is, the difference between the volumetric efficiency and unity, could only be due to a rise in temperature of the charge. The indicator diagrams of the present report (figs. 9(a) and 9(b)) confirm the hypothesis of reference 1, that under the conditions of operation in this report and in reference 1 the volumetric deficiency can be completely explained by the rise in temperature of the charge due to heat transfer and due to

work done in forcing the charge through the inlet system.

2. Reducing the flow resistance of the inlet valve may be expected to reduce the charge temperature by decreasing the amount of work done in forcing the charge through the valve. Since heat transfer is closely related to friction, it is probable that a reduction in resistance will also reduce the heat transfer at this point. Since turbulence in the cylinder will also be reduced, it is not clear whether the detonation tendency will be reduced or increased by decreasing inlet-valve resistance.

3. Inlet-valve temperatures under engine operating conditions ranged from 642° F without inlet-valve and seat cooling to 422° F with full inlet-valve and seat cooling at 1500 rpm. At 1000 rpm the average temperature was from 620° F to 393° F.

4. The total rise in temperature of the charge between the inlet tank and the cylinder after the inlet valve was closed was 81° F for 1500 rpm and 104° F for 1000 rpm under normal full-throttle operating conditions (without inlet-valve and seat cooling).

5. With this particular cylinder the volumetric efficiency expressed in percentage will be increased by 0.10 percent for each 10° F drop in the average temperature of the inlet valve and seat at 1500 rpm and 0.15 percent at 1000 rpm.

6. The temperature rise of the charge under normal operating conditions due to heat transfer from the inlet-valve and seat to the flowing charge was 34° F at 1500 rpm or 42 percent of the total rise. At 1000 rpm the rise was 36° F, which was 35 percent of the total rise.

7. The temperature rise under normal operating conditions resulting from pressure drop through the inlet valve was 26° F at 1500 rpm and 12° F at 1000 rpm.

8. The unaccounted-for temperature rise under normal operating conditions, including the temperature rise of the charge during the time it lies behind the closed intake valve and the rise after the charge is in the cylinder, was 20° F at 1500 rpm and 56° F at 1000 rpm.

9. Each 10° F reduction in the average temperature of the inlet valve and seat will allow opening the throttle to obtain approximately 0.7-percent increase in indicated power with constant tendency to detonate.

Massachusetts Institute of Technology,
Cambridge, Mass., September 10, 1941.

APPENDIX I

COMPUTATION OF TEMPERATURE-RISE RATIO

In the flow model the rate at which heat is transferred to the working fluid is

$$\frac{dQ}{dt} = \Delta_2 t \ w \ c_p = \phi \ \Delta_1 t \ w \ c_p \tag{1}$$

The conditions in the flow model are similar to those of the case considered in reference 6 (pp. 128-132) that is, the surfaces of the port and the inlet valve are at a substantially uniform temperature. The analysis of this reference therefore applies, and there may be written

$$\frac{dQ}{dt} = l^2 \ K_2 \ c_p \ (\rho u)^n \left(\frac{l}{\mu}\right)^{n-1} \tag{2}$$

where the undefined symbols have the following significance:

u velocity of working fluid through inlet valve and port

μ viscosity of working fluid

l characteristic dimension of inlet valve and port

n an exponent depending on shape and orientation of inlet valve and port and subject to experimental evaluation

K_2 a nondimensional constant

Combining equations (1) and (2) and solving for ϕ gives

$$\phi = \frac{K_2 \ (\rho u)^n \left(\frac{l}{\mu}\right)^{n-1} l^2}{w} \tag{3}$$

Now

$$w = K_3 \ \rho u \ l^2 \tag{4}$$

where K_3 is a constant depending on the geometry of the inlet system.

Combining (3) and (4):

$$\phi = K_4 \, (\rho u)^{n-1} \left(\frac{l}{\mu}\right)^{n-1} \tag{5}$$

For fresh mixture and for air, respectively,

$$\phi_m = K_4 \, (\rho_m u_m)^{n-1} \left(\frac{l}{\mu_m}\right)^{n-1} \tag{6}$$

$$\phi_a = K_4 \, (\rho_a u_a)^{n-1} \left(\frac{l}{\mu_a}\right)^{n-1} \tag{7}$$

For the same mass rate of flow of fresh charge and air:

$$\rho_m u_m = \rho_a u_a \tag{8}$$

Combining (5), (6), and (7):

$$\phi_m = \phi_a \left(\frac{\mu_a}{\mu_m}\right)^{n-1} \tag{9}$$

For a mixture of air and octane the viscosity will be approximately proportional to the sum of the molar fractions of the constituent viscosities (private communication from Prof. F. G. Keyes to Prof. E. S. Taylor), that is:

$$\mu_m = \frac{\mu_a \dfrac{W_a}{m_a}}{\dfrac{W_a}{m_a} + \dfrac{W_f}{m_f}} + \frac{\mu_f \dfrac{W_f}{m_f}}{\dfrac{W_a}{m_a} + \dfrac{W_f}{m_f}} \tag{10}$$

the subscript f refers to the fuel and m is the molecular weight of a constituent of the mixture.

Calculation of μ_m made on the basis of equation (10) using the estimated viscosity of octane 0.000100 poise (communication from Prof. F. G. Keyes) and that of air 0.000218 poise gives at room temperature the value $\mu_m = 0.000215$ poise.

From the flow-model data the average value of n was found to be 0.74.

Inserting these values in equation (9) gives the relationship:

$$\Phi_m = 0.997\ \Phi_a$$

APPENDIX II

DERIVATION OF EQUATION FOR $T_c - T_i$

Equation (5) as given in the text is:

$$(M_f + M_r)E_c - M_f H_i - M_r E_r = -\frac{1}{J}\int_e^c P dv + Q_{ec} \tag{5}$$

Rearranging:

$$\frac{M_r}{M_f}(E_c - E_r) + E_c - H_i = -\frac{1}{JM_f}\int_e^c P dv + \frac{Q_{ec}}{M_f} \tag{6}$$

$$\frac{M_r}{M_r + M_f} = f \tag{7}$$

$$\frac{M_r}{M_f} = \frac{f}{1 - f} \tag{8}$$

$$E_c = H_c - \frac{P_c V_c}{J} \tag{9}$$

Combining equations (6), (8), and (9):

$$\frac{f}{1 - f}(E_c - E_r) + H_c - H_i = \frac{P_c V_c}{J} - \frac{1}{JM_f}\int_e^c P dv + \frac{Q_{ec}}{M_f} \tag{10}$$

If the residual and fresh charges are kept separated by a membrane:

$$H_c(M_f + M_r) = M_r H_{cr} + M_f H_{cf} \tag{11}$$

Hence:

$$H_c = \frac{M_r}{M_r + M_f} H_{cr} + \frac{M_f}{M_f + M_r} H_{cf} \tag{12}$$

and from (5):

$$E_c = \frac{M_r}{M_r + M_f} H_{cr} + \frac{M_f}{M_f + M_r} H_{cf} - \frac{P_c V_c}{J} \tag{13}$$

and from (7):

$$E_c = f(H_{cr}) + (1 - f)H_{cf} - \frac{P_c V_c}{J} \tag{14}$$

$$H_c = f(H_{cr}) + (1 - f)H_{cf} \tag{15}$$

Combining (10), (14), and (15):

$$\frac{f}{1 - f}\left[fH_{cr} + (1-f)H_{cf} - \frac{1}{J} P_c V_c - E_r\right] + fH_{cr} + (1-f)H_{cf} - H_i$$

$$= \frac{1}{J} P_c V_c - \frac{1}{JM_f}\int_e^c P dv + \frac{Q_{ec}}{M_f} \tag{16}$$

Rearranging (16):

$$\frac{f}{1-f}(H_{cr} - E_r) - \left(\frac{f}{1-f}\right)\frac{P_c V_c}{J} + H_{cf} - H_i$$

$$= \frac{1}{J} P_c V_c - \frac{1}{JM_f}\int_e^c P dv + \frac{Q_{ec}}{M_f} \qquad (17)$$

But

$$H_{cr} = E_{cr} + \frac{P_{cr} V_{cr}}{J}$$

Hence

$$\frac{f}{1-f}(E_{cr} - E_r) - \left(\frac{f}{1-f}\right)\frac{P_c V_c - P_{cr} V_{cr}}{J} + H_{cf} - H_i$$

$$= \frac{1}{J} P_c V_c - \frac{1}{JM_f}\int_e^c P dv + \frac{Q_{ec}}{M_f} \qquad (18)$$

Combining like terms and allowing that $P_{cr} = P_c$:

$H_{cf} - H_i$

$$= \frac{P_c}{J}\left[\frac{1}{1-f}V_c - \frac{f}{1-f}V_{cr}\right] - \frac{f}{1-f}(E_{cr} - E_r) - \frac{1}{JM_f}\int_e^c P dv + \frac{Q_{ec}}{M_f} \qquad (19)$$

Now

$$M_r V_{cr} + M_f V_{cf} = (M_r + M_f) V_c \qquad (20)$$

(Sum of partial volumes equals total volume, if $P_c = P_{cr}$)

Combining (7), (8), and (20):

$$V_{cf} = \frac{1}{1-f}V_c - \frac{f}{1-f}V_{cr} \qquad (21)$$

Substituting (21) in (19):

$$H_{cf} - H_i$$

$$= \frac{1}{J}\left[(P_c V_{cf}) - \frac{1}{M_f}\int_e^c Pdv\right] - \frac{f}{1-f}(E_{cr} - E_r) + \frac{Q_{ec}}{M_f} \quad (22)$$

Multiplying the bracket term in (22) by $\frac{M_f}{M_f}$:

$$H_{cf} - H_i$$

$$= \frac{1}{JM_f}\left[P_c M_f V_{cf} - \int_e^c Pdv\right] - \frac{f}{1-f}(E_{cr} - E_r) + \frac{Q_{ec}}{M_f} \quad (23)$$

Now $$M_f V_{cf} = v_c - v_{cr} \quad (24)$$

and $$\frac{v_{cr}}{v_{er}} = \left(\frac{P_{er}}{P_{cr}}\right)^{\frac{1}{k}} = \left(\frac{P_e}{P_c}\right)^{\frac{1}{k}} \quad \text{since } P_{cr} = P_c \text{ and } P_{er} = P_e \quad (25)$$

Combining (23), (24), and (25):

$$H_{cf} - H_i = \frac{1}{JM_f}\left[P_c v_c - P_c v_{er} + P_c v_{er}\left(1 - \left(\frac{P_{er}}{P_c}\right)^{\frac{1}{k}}\right) - \int_e^c Pdv\right]$$

$$- \frac{f}{1-f}(E_{cr} - E_r) + \frac{Q_{ec}}{M_f} \quad (26)$$

From the assumption of adiabatic conditions for the residual:

$$E_{cr} - E_r = \frac{1}{JM_r}\int_{P_{er}}^{P_{cr}} P dv = \frac{P_{er}V_{er}}{J(k-1)}\left[\left(\frac{P_{cr}}{P_{er}}\right)^{\frac{k-1}{k}} - 1\right] \tag{27}$$

$$V_{er} = \frac{v_{er}}{M_r} \tag{28}$$

Combining (8), (26), (27), and (28), and noting that $P_{cr} = P_c$:

$$H_{cf} - H_i = \frac{1}{JM_f}\left[(P_c v_c - P_c v_{er}) - \int_e^c P dv\right]$$

$$+ \frac{V_{er}f}{J(1-f)}\left[P_c\left(1-\left(\frac{P_{er}}{P_c}\right)^{\frac{1}{k}}\right) + \frac{P_{er}}{k-1}\left(1-\left(\frac{P_c}{P_{er}}\right)^{\frac{k-1}{k}}\right)\right]$$

$$+ \frac{Q_{ec}}{M_f} \tag{29}$$

$$M_f = (1 + f)M \tag{30}$$

Combining (8), (28), and (30):

$$V_{er} = \frac{1-f}{f(1+f)}\frac{v_{er}}{M} \tag{31}$$

Combining (29), (30), and (31):

$$H_{cf} - H_i = \frac{1}{J(1 + f)M} \left[(P_c v_c - P_c v_{er}) - \int_e^c P dv \right]$$

$$+ \frac{v_{er}}{J(1 + f)M} \left[P_c \left(1 - \left(\frac{P_{er}}{P_c} \right)^{\frac{1}{k}} \right) + \frac{P_{er}}{k-1} \left(1 - \left(\frac{P_c}{P_{er}} \right)^{\frac{k-1}{k}} \right) \right]$$

$$+ \frac{Q_{ec}}{(1 + f)M} \qquad (32)$$

Now

$$H_{cf} - H_i = (T_{cf} - T_i)\, c_p$$

where T_{cf} is the temperature of the fresh charge at point c before mixing with the residual gas. Also, v_{er} is the clearance volume v_1.

$$T_{cf} - T_i = \frac{Q_{ec}}{c_p(1+f)M} + \frac{1}{c_p J(1+f)M} \left[(P_c v_c - P_c v_1) - \int_e^c P dv \right]$$

$$+ \frac{v_1}{c_p J(1+f)M} \left\{ P_c \left[1 - \left(\frac{P_{er}}{P_c} \right)^{\frac{1}{k}} \right] + \frac{P_{er}}{k-1} \left[1 - \left(\frac{P_c}{P_{er}} \right)^{\frac{k-1}{k}} \right] \right\} \qquad (33)$$

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TABLE II

COMPARISON OF RESULTS OF INTEGRATED AND SHORT-METHOD CALCULATIONS FOR THE HEAT Q TRANSFERRED FROM THE INTAKE VALVE AND SEAT

Short method $Q = \Delta_2 t\ W_a\ c_p$ (Btu)	Integration $Q = \Sigma q' \Delta_2 t\ c_p$ (Btu)	Percentage error
0.0447	0.0442	1.1
.0434	.0432	.6
.0372	.0359	3.5
.0357	.0346	2.9
.0219	.0213	2.9
.0281	.0274	2.8
.0452	.0442	2.3
.0424	.0427	.6

Table I

Integration of the Heat, Q, Transferred from the Intake-Valve and Seat

[Run 6, fig. 6]

Lift, (in.)	$\Delta\theta$ (deg)	ΔT (seconds)	ΔP (in. Hg)	W_a (lb/sec)	q (lb)	W_a' (lb/sec)	q' (lb)	ϕ_a	Δt (°F)	$\Delta_{\phi} t$ (°F)	ΔQ (Btu)
1/16	16	0.00267	0.25	0.0187	0.0000499	0.0138	0.0000368	0.366	225	82.4	0.00076
1/8	10	.00167	.58	.0510	.0000852	.0376	.0000629	.247	225	55.6	.00088
3/16	8	.00134	.73	.0824	.000110	.0608	.0000815	.173	225	38.9	.00080
1/4	10	.00167	.75	.109	.000182	.0804	.000134	.158	225	35.6	.00120
5/16	10	.00167	.75	.125	.000209	.0923	.000154	.151	225	34.0	.00132
3/8	12	.00201	.65	.127	.000255	.0937	.000188	.150	225	33.8	.00160
7/16	18	.00301	.68	.130	.000391	.0959	.000289	.149	225	33.5	.00243
1/2	25	.00417	1.43	.189	.000788	.139	.000582	.129	225	29.0	.00423
1/2	25	.00417	2.05	.222	.000926	.164	.000683	.119	225	26.8	.00460
7/16	16	.00267	2.05	.222	.000593	.164	.00438	.119	225	26.8	.00294
3/8	14	.00234	1.70	.205	.000480	.151	.000354	.125	225	28.1	.00250
5/16	10	.00167	1.00	.145	.000242	.107	.000179	.143	225	32.2	.00149
1/4	10	.00167	.45	.0843	.000141	.0622	.000104	.172	225	38.7	.00101
3/16	6	.00100	.78	.0854	.0000854	.0630	.0000630	.171	225	38.5	.00061
1/8	10	.00167	.60	.0519	.0000867	.0383	.0000640	.245	225	55.1	.00089
1/16	2	.000334	.15	.0148	.0000049	.0109	.0000036	.387	225	87.1	.00078
	$\Sigma\Delta\theta$ = 202	$\Sigma\Delta T$ = 0.0337			Σq = 0.00463						$\Sigma\Delta Q$ = 0.0274

Table III

Experimental Values of Heat Q Transferred from the Valve

and Seat Obtained in Wright J-6 Engine Cylinder Tests

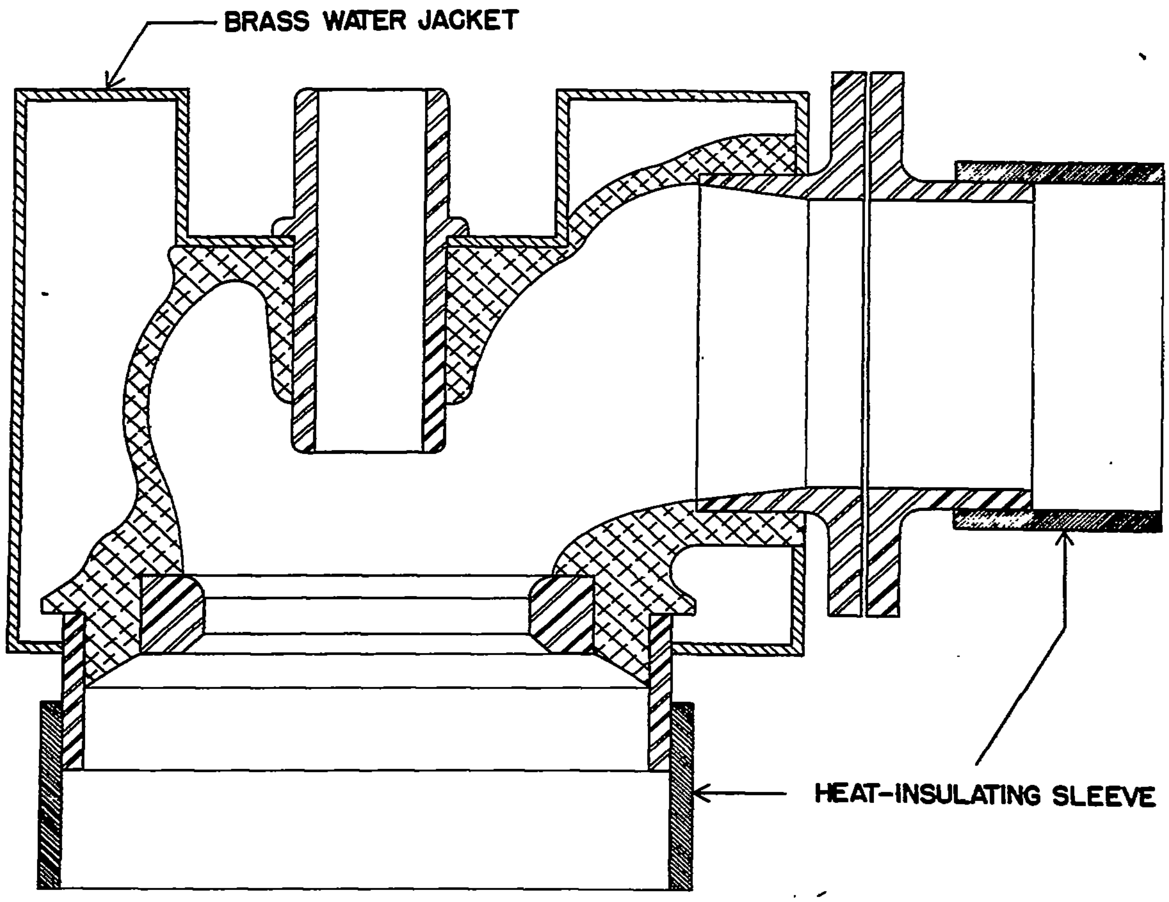
(Computed values based on short-cut method of calculations)

Run	Engine speed (r p m)	t_p (°F)	w_a (lb/sec)	M_m (lb/stroke)	w'_m (lb/sec)	ϕ'_m	t_1 (°F)	t_v (°F)	t_s (°F)	t_{vs} (°F)	$\Delta_1 t$ (°F)	$\Delta_2 t$ (°F)	Q (B t u)
30	1000	402	0.0298	0.00387	0.115	0.140	117	533	430	482	365	51.1	0.0454
31	1000	400	.0310	.00403	.119	.137	119	393	221	307	188	25.8	.0239
32	1000	400	.0307	.00398	.118	.138	118	438	233	336	218	30.1	.0276
33	1000	400	.0309	.00402	.119	.137	117	440	303	372	255	34.9	.0323
34	1000	400	.0300	.00390	.116	.139	118	516	418	467	349	48.5	.0435
38	1000	400	.0295	.00381	.114	.140	119	620	430	525	406	56.8	.0498
40	1000	401	.0306	.00397	.119	.137	117	432	215	324	207	28.4	.0259
41	1000	402	.0306	.00397	.119	.137	117	500	300	400	283	38.8	.0354
43	1000	403	.0300	.00390	.116	.139	118	619	436	528	410	57.0	.0512
44	1000	400	.0304	.00395	.117	.139	119	512	417	465	346	48.2	.0438
45	1500	400	.0468	.00405	.181	.113	119	633	434	534	415	46.8	.0437
46	1500	401	.0476	.00412	.184	.112	120	538	417	478	358	40.2	.0381
47	1500	400	.0483	.00418	.186	.111	119	422	227	325	206	22.9	.0220
48	1500	400	.0483	.00418	.186	.111	118	475	231	353	235	26.1	.0252
49	1500	400	.0477	.00413	.185	.112	118	642	428	535	417	46.7	.0443
50	1500	400	.0473	.00410	.179	.115	118	641	428	535	417	47.8	.0452
51	1500	400	.0482	.00417	.186	.111	119	528	302	415	296	32.9	.0315
52	1500	400	.0482	.00417	.186	.111	118	457	296	377	259	28.8	.0276
53	1500	400	.0482	.00417	.186	.111	118	494	310	402	284	31.5	.0302
54	1500	400	.0475	.00412	.184	.112	118	639	433	536	418	46.8	.0443

Table IV

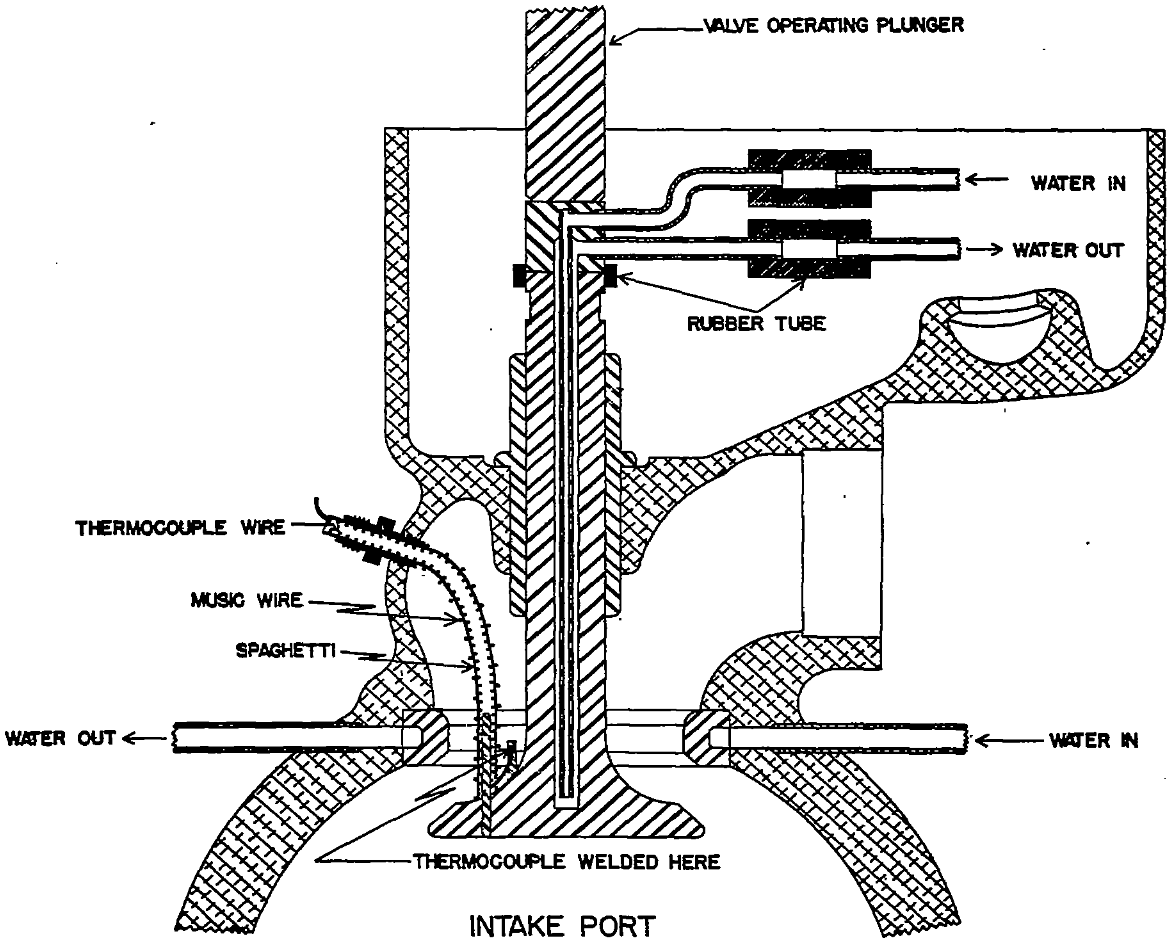
Experimental Values of Over-All Heat Q_T Obtained in Wright J-6 Engine Cylinder Tests

Run	Engine speed (r p m)	ρ_i (lb/ft^3)	P_e (in.Hg)	P_i (in.Hg)	P_e/P_i	[illegible]	W_a (lb/sec)	T_c (°F abs)	T_i (°F abs)	ΔT_c (°F)	t_p (°F)	t_s (°F)	t_v (°F)	t_{vs} (°F)	M_a (lb/stroke)	Q (B t u)	Q_T (B t u)	Q/Q_T x 100	Cooling
30	1000	0.0740	30.01	30.35	0.988	0.838	0.0323	679	577	102	402	430	533	482	0.00387	0.0454	0.0908	50.0	Valve
31	1000	.0738	30.01	30.37	.988	.873	.0336	656	579	77	400	221	393	307	.00403	.0239	.0714	33.5	Valve & seat, both full H_2O
32	1000	.0738	30.01	30.35	.988	.864	.0332	662	578	84	400	233	438	336	.00398	.0276	.0772	35.8	Valve & seat; less H_2O in seat than (31)
33	1000	.0740	30.01	30.35	.988	.868	.0335	657	577	80	400	303	440	372	.00402	.0323	.0739	43.7	Valve & seat; less H_2O than (32)
34	1000	.0739	30.01	30.37	.988	.845	.0325	677	578	99	400	418	516	467	.00390	.0435	.0888	48.9	Valve as (30)
38	1000	.0737	30.01	30.35	.988	.833	.0319	688	579	109	400	430	620	525	.00381	.0498	.0961	51.8	None
40	1000	.0740	30.01	30.35	.988	.860	.0331	663	577	86	401	215	432	324	.00397	.0259	.0786	33.0	Seat full H_2O
41	1000	.0740	30.01	30.35	.988	.860	.0331	663	577	86	402	300	500	400	.00397	.0354	.0786	45.2	Seat, less H_2O than (36)
43	1000	.0738	30.01	30.35	.988	.845	.0325	677	578	99	403	436	619	528	.00390	.0512	.0888	57.7	None
44	1000	.0737	30.01	30.35	.988	.860	.0329	666	579	87	400	417	512	465	.00395	.0438	.0792	55.3	Valve as (30)
45	1500	.0724	30.89	29.89	1.03	.897	.0507	668	579	89	400	434	633	534	.00405	.0437	.0831	52.6	None
46	1500	.0723	30.89	29.89	1.03	.903	.0515	663	580	83	401	417	538	478	.00412	.0381	.0787	48.4	Valve full H_2O
47	1500	.0724	30.89	29.89	1.03	.925	.0523	647	579	68	400	227	422	325	.00418	.0220	.0654	33.6	Valve & seat, full H_2O
48	1500	.0726	30.89	29.89	1.03	.922	.0523	648	578	70	400	231	475	353	.00418	.0252	.0673	37.5	Seat full H_2O
49	1500	.0726	30.89	29.89	1.03	.912	.0517	653	578	75	400	428	642	535	.00413	.0443	.0713	62.2	None
50	1500	.0728	30.89	29.96	1.03	.903	.0513	659	578	81	400	428	641	535	.00410	.0452	.0763	59.3	None
51	1500	.0727	30.89	29.96	1.03	.921	.0522	648	579	69	400	302	528	415	.00417	.0315	.0663	47.8	Seat, less H_2O than (48)
52	1500	.0728	30.89	29.96	1.03	.920	.0522	648	578	70	400	296	457	377	.00417	.0276	.0673	41.2	Seat as (51); valve full H_2O
53	1500	.0728	30.89	29.96	1.03	.920	.0522	648	578	70	400	310	494	402	.00417	.0302	.0673	45.0	Seat, reduced H_2O; valve, reduced H_2O
54	1500	.0728	30.89	29.96	1.03	.906	.0514	658	578	80	400	433	639	536	.00412	.0449	.0758	58.6	None



SECTION OF WATER-JACKETED FLOW MODEL.

FIGURE 1.



INTAKE PORT
SHOWING ARRANGEMENTS FOR COOLING
VALVE AND SEAT AND MEASURING VALVE TEMPERATURE.

FIGURE 2.

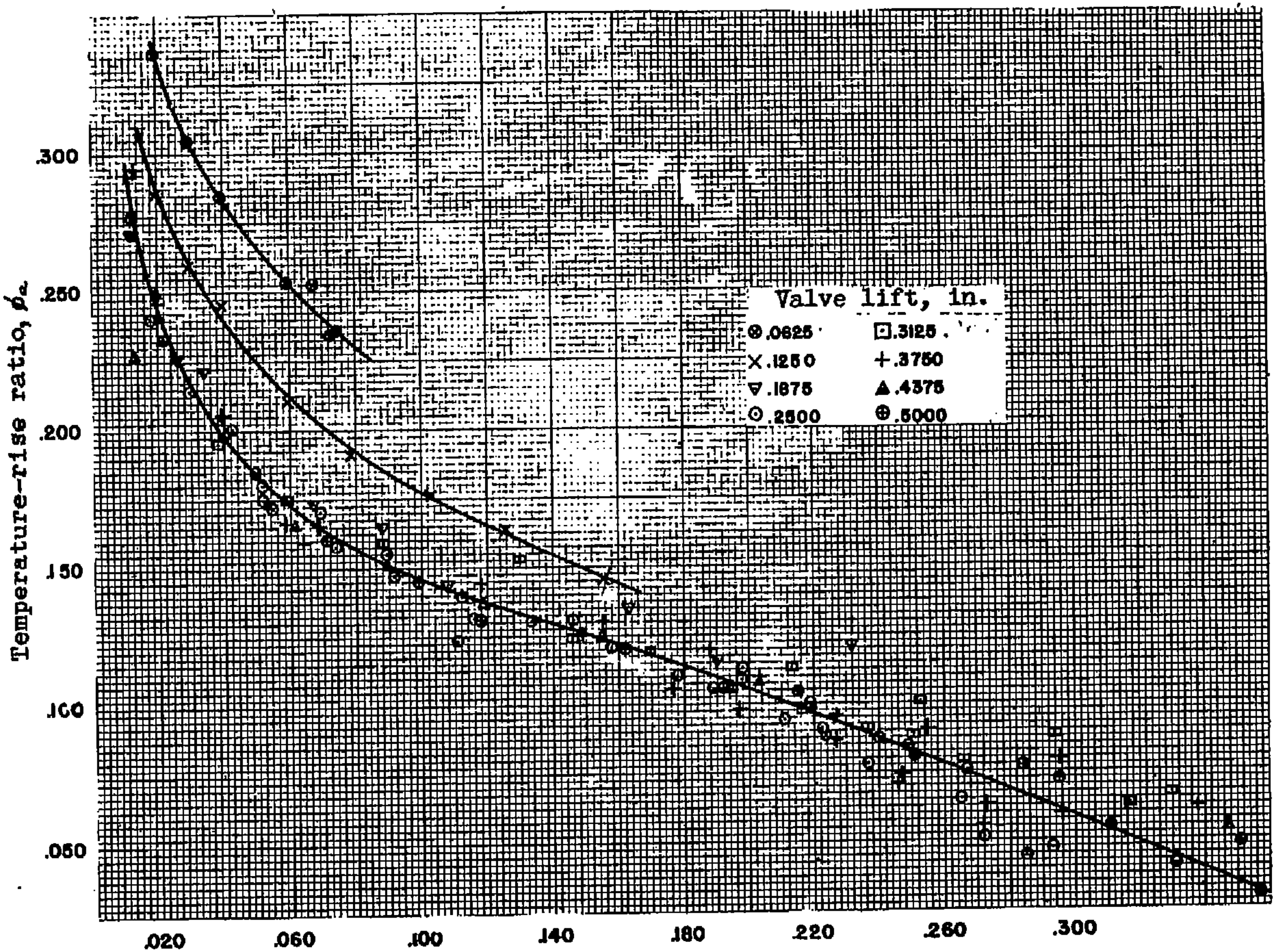


Figure 3.- Variation of temperature-rise ratio with air flow.

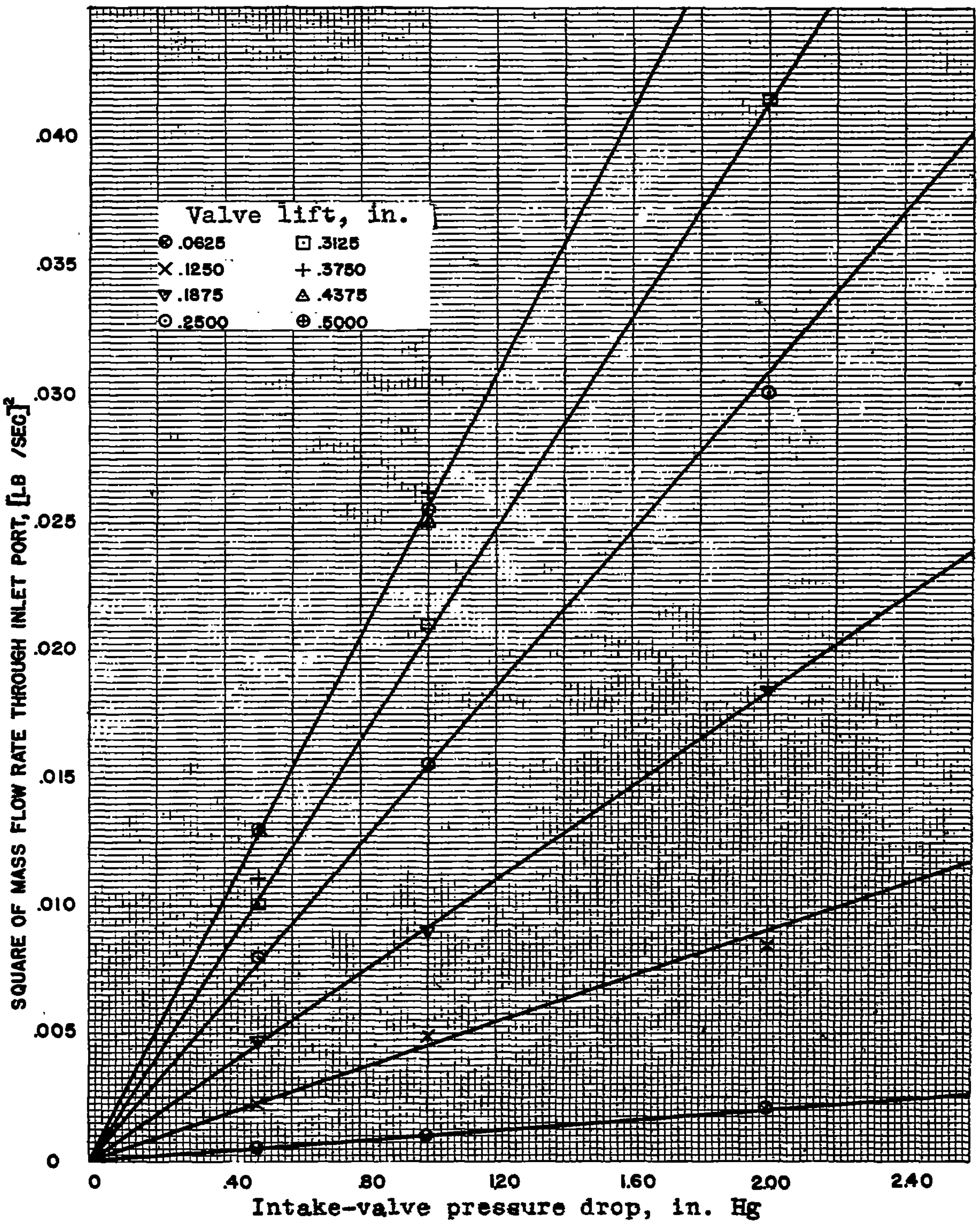


Figure 4.- Variation of mass flow rate with intake-valve pressure drop.

ENGINE CYLINDER HEAD

SHOWING FLEXIBLE INLET-VALVE COOLING WATER CONNECTIONS

AND THERMOCOUPLE LEADS TO VALVE AND SEAT.

FIGURE 5.

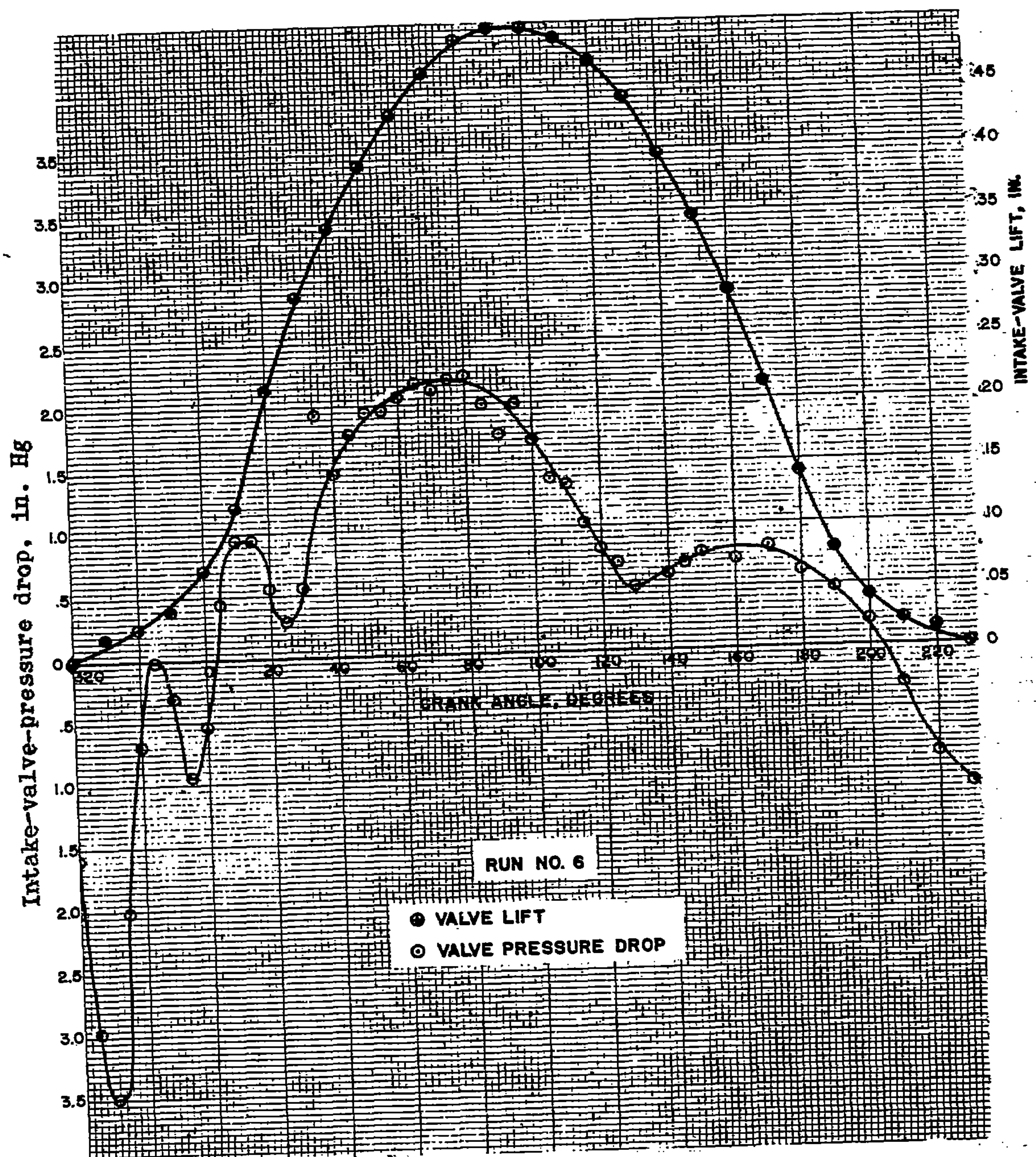


Figure 6.- Variation of intake-valve pressure drop and lift with crank angle. Run 6.

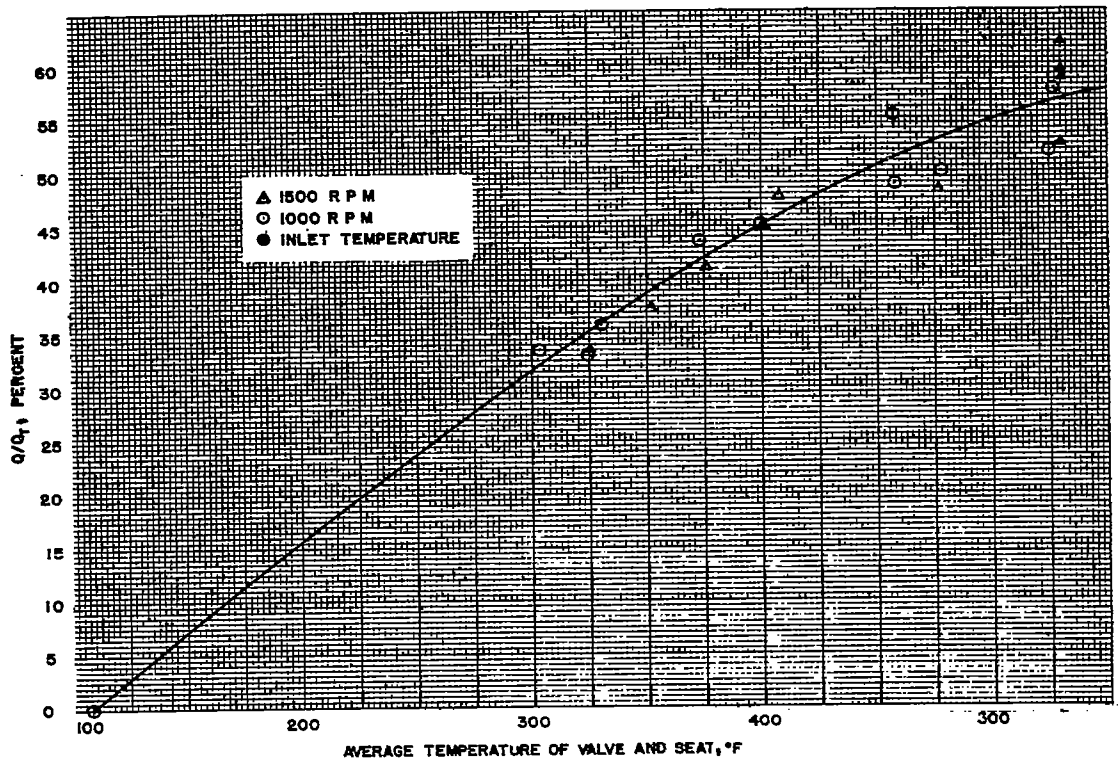


FIGURE 8.

VARIATION OF Q/Q₁ WITH VALVE AND SEAT TEMPERATURE.

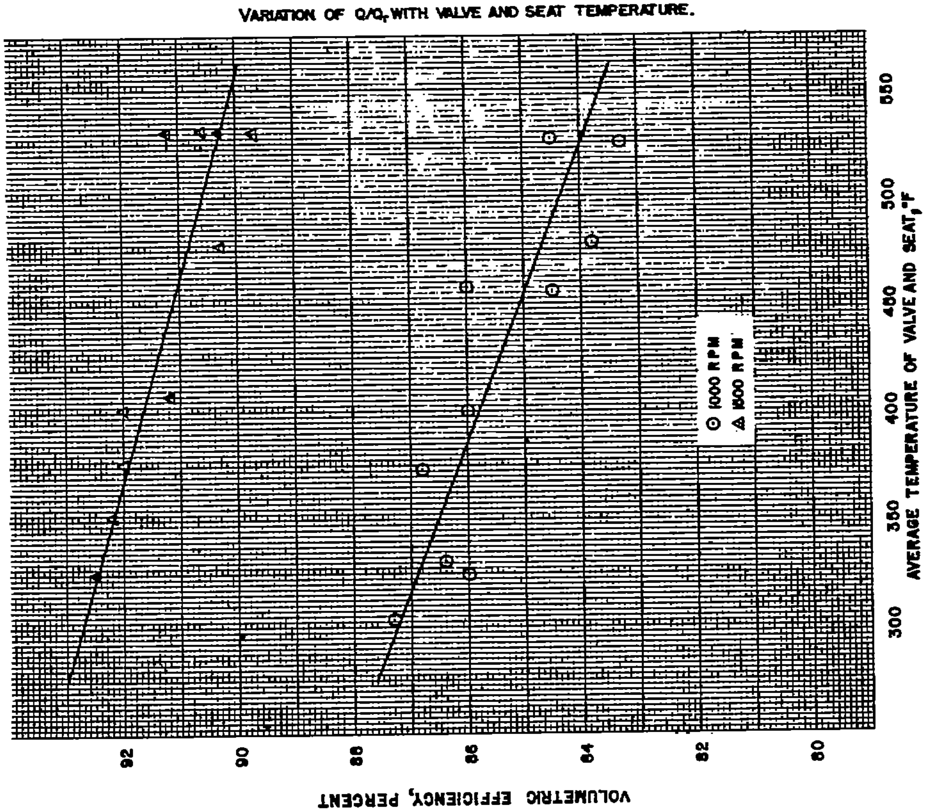


FIGURE 7.

VARIATION OF VOLUMETRIC EFFICIENCY WITH VALVE AND SEAT TEMPERATURE.

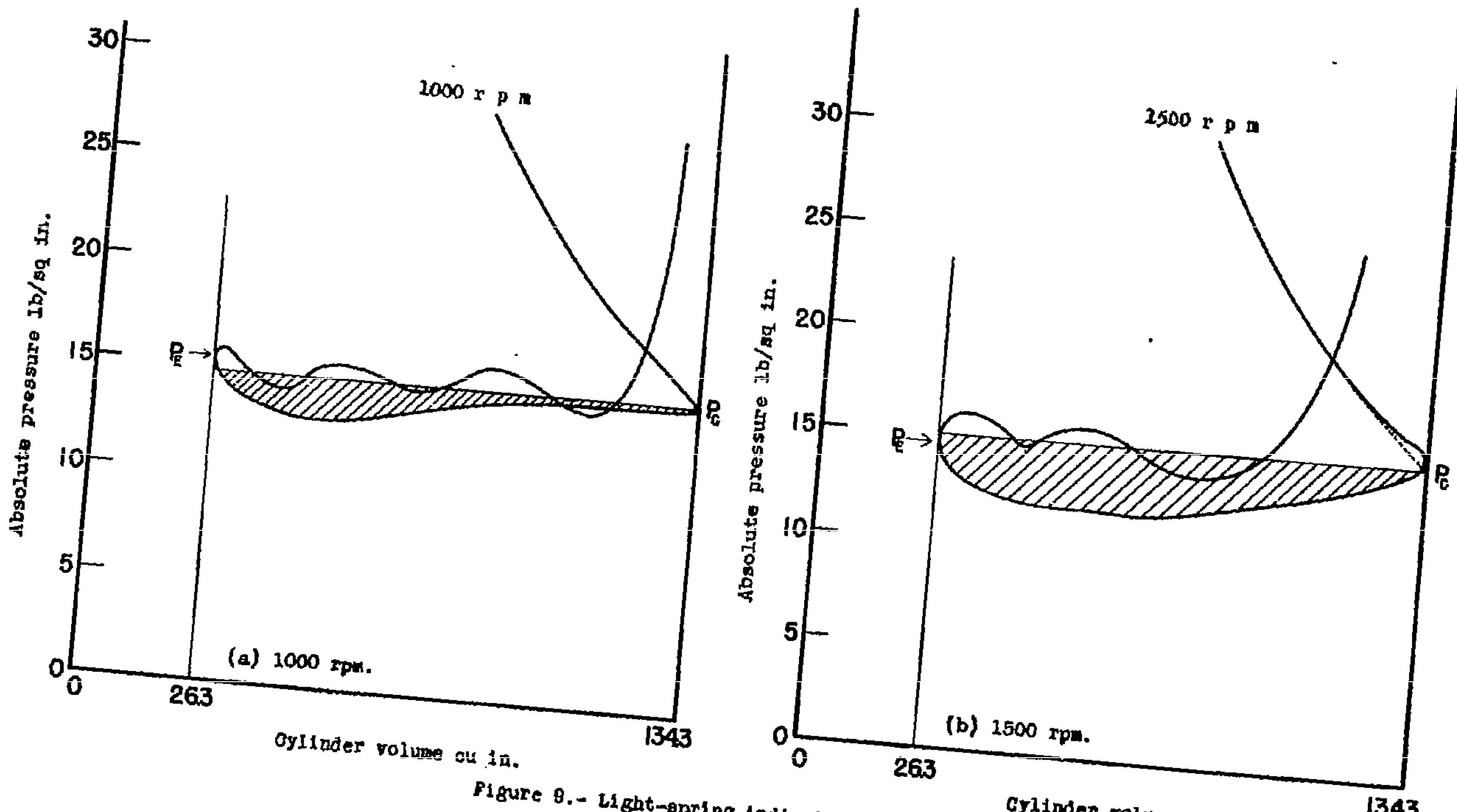


Figure 9.- Light-spring indicator card.